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(54) Closed loop launch and creep control for automatic clutch

Start- und Kriechregelung für automatische Kupplung

Régulation de démarrage et de rampage pour un embrayage automatique

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Description

Technical Field of the Invention

The technical field of this invention is that of automatic clutch controls, and more particularly closed loop automatic clutch controls for reducing oscillatory response to launch and creep of a motor vehicle.

Background of the Invention

In recent years there has been a growing interest in increased automation in the control of the drive train of motor vehicles, and most especially in control of the drive train of large trucks. The use of automatic transmissions in passenger automobiles and light trucks is well known. The typical automatic transmission in such a vehicle employs a fluid torque converter and hydraulically actuated gears for selecting the final drive ratio between the engine shaft and the drive wheels. This gear selection is based upon engine speed, vehicle speed and the like. It is well known that such automatic transmissions reduce the effectiveness of the transmission of power from the engine to the drive shaft, with the consummate reduction in fuel economy and power as compared with the skilled operation of a manual transmission. Such hydraulic automatic transmissions have not achieved wide spread use in large motor trucks because of the reduction in efficiency of the operation of the vehicle.

One of the reasons for the loss of efficiency when employing a hydraulic automatic transmission is loss occurring in the fluid torque converter. A typical fluid torque converter exhibits slippage and consequent loss of torque and power in all modes. It is known in the art to provide lockup torque converters that provide a direct link between the input shaft and the output shaft of the transmission above certain engine speeds. This technique provides adequate torque transfer efficiency when engaged, however, this technique provides no gain in efficiency at lower speeds.

It has been proposed to eliminate the inefficiencies inherent in a hydraulic torque converter by substitution of an automatically actuated friction clutch. This substitution introduces another problem not exhibited in the use of the hydraulic torque converters. The mechanical drive train of a motor vehicle typically exhibits considerable torsional compliance in the driveline between the transmission and the traction wheels of the vehicle. This torsional compliance may be found in the drive shaft between the transmission and the differential or the axle shaft between the differential and the driven wheels. It is often the case that independent design criteria encourages or requires this driveline to exhibit considerable torsional compliance. The existence of substantial torsional compliance in the driveline of the motor vehicle causes oscillatory response to clutch engagement. These oscillatory responses can cause considerable additional wear to the drive train components and other parts of the vehicle. In addition, these oscillatory responses can cause objectionable passenger compartment vibrations.

The oscillatory response of the driveline to clutch engagement is dependent in large degree to the manner in which the input speed of the transmission, i.e. the speed of the clutch, approaches the engine speed. A smooth approach of these speeds, such as via a decaying exponential function, imparts no torque transients on clutch lockup. If these speeds approach abruptly, then a torque transient is transmitted to the driveline resulting in an oscillatory response in the vehicle driveline.

Thus it would be an advantage to provide automatic clutch actuation of a friction clutch that reduces the oscillatory response to clutch engagement.

GB-A-2231116 discloses an apparatus for controlling an automatic friction clutch which generates a differential speed corresponding to actual slip as measured by sensors detecting the speed of the engine and the speed of rotation of a gear input shaft. This detected differential speed or actual slip is compared with an ideal slip speed stored in a processor which periodically examines input parameters, such as clutch position, throttle valve position, engine temperature and gear position to generate the ideal slip based on the specific operating state of the vehicle. The difference between the actual slip and the ideal slip is calculated and used to control a clutch actuator to bring the actual contact pressure of the clutch or the actual clutch position to an ideal contact pressure or clutch position to achieve the ideal slip speed.

The problem of providing such automatic clutch actuation is considerably increased in large trucks. In particular, large trucks exhibit a wide range of variability in response between trucks and within the same truck. The total weight of a particular large truck may vary over an 8 to 1 range from unloaded to fully loaded. The driveline compliance may vary over a range of about 2 to 1 among different trucks. Further, the clutch friction characteristic may vary within a single clutch as a function of degree of clutch engagement and between clutches. It would be particularly advantageous to provide such an automatic clutch actuation system that does not require extensive adjustment to a particular motor vehicle or the operating condition of the motor vehicle.

Summary of the Invention

This invention is an automatic clutch controller used in a combination including a source of motive power, a friction clutch, and at least one inertially-loaded traction wheel connected to the friction clutch that has a torsional compliance exhibiting an oscillatory response to torque inputs. The automatic clutch controller is preferably used with a transmission shift controller. This automatic clutch controller provides smooth clutch engagement during vehicle launch, following transmission shifts and during creep to minimize the oscillatory response to clutch engagement. This automatic clutch controller is useful in large trucks.

The automatic clutch controller receives inputs from an engine speed sensor and a transmission input speed sensor. The transmission input speed sensor senses the rotational speed at the input to the transmission, which is the output of the friction clutch. The automatic clutch controller develops a clutch engagement signal controlling a clutch actuator between fully disengaged and fully engaged. The clutch engagement signal engages the friction clutch in a manner causing asymptotic approach of the transmission input speed to a reference speed. This minimizes the oscillatory response to torque inputs of the inertially-loaded traction wheel.

In the preferred embodiment the automatic clutch controller operates in two modes. In a launch mode, corresponding to normal start of the vehicle, the clutch engagement signal causes the transmission input speed to asymptotically approach the engine speed. This same mode may optionally also be used for clutch re-engagement upon transmission gear shifts. In a creep mode, corresponding to slow speed creeping of the vehicle, the clutch engagement signal causes the transmission input speed to asymptotically approach a creep reference signal. This creep reference signal is generated based on the amount of throttle and the engine speed. The two modes are selected based upon the throttle setting. The launch mode is selected for a throttle of more than 25% full throttle, otherwise the creep mode is selected.

The automatic clutch controller includes construction to reduce the need for detailed particularization for individual vehicles or vehicle models. A transmission input speed reference signal is supplied to a prefilter. This transmission input speed reference signal corresponds to the engine speed when the launch mode is selected and the creep reference signal when the creep mode is selected. The prefilter serves to shape the system transient response. An algebraic summer forms the controlled error by subtracting the transmission input speed signal from the prefiltered transmission input speed reference signal. This error signal is supplied to a compensator having sufficient gain as a function of frequency to reduce the system closed loop sensitivity to vehicle parameter variations. The compensator produces a clutch engagement signal for controlling clutch engagement in a manner to minimize the oscillatory response to clutch engagement.

The automatic clutch controller is preferably implemented in discrete difference equations executed by a digital microcontroller. The microcontroller implements a compensator having a transfer function approximately the inverse of the transfer function of the inertially-loaded traction wheel. This compensator transfer function includes a notch filter covering the region of expected oscillatory response of the driveline. The frequency band of this notch filter must be sufficiently broad to cover a range of frequencies because the oscillatory response frequency may change with changes in vehicle loading and driveline characteristics. The compensator also preferably provides an elevated response in range of frequencies where the driveline response is a minimum to increase the loop gain and reduce sensitivity to variations in vehicle characteristics.

The clutch actuation controller preferably stores sets of coefficients for the discrete difference equations corresponding to each gear ratio of the transmission. The clutch actuation controller recalls the set of coefficients corresponding to the selected gear ratio. These recalled set of coefficients are employed in otherwise identical discrete difference equations for clutch control.

The controller preferably includes an integral error function for insuring full clutch engagement within a predetermined interval of time after initial partial engagement when in the launch mode. Any long term difference between the transmission input speed reference signal and the transmission input speed eventually drives the clutch to full engagement. The controller preferably also includes a second integral function to ensure clutch lockup even if the engine speed in increasing.

The integral function and the second integral function are preferably disabled when the rate of engine speed increase falls below a predetermined threshold. This level could be zero, disabling the first and second integral functions when the engine speed decreases. A threshold detector determines when to disable the integrators based on the differential signal. Respective switches connected to the threshold detector enables and disables integration. This permits delay of the advance of the clutch when the rate of engine speed increase falls below the threshold. This will generally occur only when accelerating under heavy load. In this case the clutch will continue to slip allowing the load to slowly accelerate until the torque demand is reduced to the available engine torque. Then the integrators will again be enabled and clutch advance will resume.

The automatic clutch controller may further include a differentiator connected to the engine speed sensor. The engine speed differential signal corresponding to the rate of change of the engine speed signal is added to the signal supplied to the clutch actuator. This differential signal causes rapid advance of clutch actuation when the engine speed

is accelerating. Rapid advance of the clutch in this case prevents the engine speed from running away. An integrator connected to the differentiator saves the clutch actuation level needed to restrain the engine speed once the engine speed is no longer accelerating.

Brief Description of the Drawings

These and other objects and aspects of the present invention will be described below in conjunction with the drawings in which:

FIGURE 1 illustrates a schematic view of the vehicle drive train including the clutch actuation controller of the present invention;
FIGURE 2 illustrates the typical relationship between clutch engagement and clutch torque;
FIGURE 3 illustrates the ideal response of engine speed and transmission input speed over time for launch of the motor vehicle;
FIGURE 4 illustrates the ideal response of engine speed and transmission input speed over time for creeping of the motor vehicle; and
FIGURE 5 illustrates a preferred embodiment of the clutch actuation controller of the present invention.

Detailed Description of the Preferred Embodiments

Figure 1 illustrates in schematic form the drive train of a motor vehicle including the automatic clutch controller of the present invention. The motor vehicle includes engine 10 as a source of motive power. For a large truck of the type to which the present invention is most applicable, engine 10 would be a diesel internal combustion engine. Throttle 11, which is typically a foot operated pedal, controls operation of engine 10 via throttle filter 12. Throttle filter 12 filters the throttle signal supplied to engine 10 by supplying a ramped throttle signal upon receipt of a step throttle increase via throttle 11. Engine 10 produces torque on engine shaft 15. Engine speed sensor 13 detects the rotational velocity of engine shaft 15. The actual site of rotational velocity detection by engine speed sensor may be at the engine flywheel. Engine speed sensor 13 is preferably a multitooth wheel whose tooth rotation is detected by a magnetic sensor.

Friction clutch 20 includes fixed plate 21 and movable plate 23 that are capable of full or partial engagement. Fixed plate 21 may be embodied by the engine flywheel. Friction clutch 20 couples torque from engine shaft 15 to input shaft 25 corresponding to the degree of engagement between fixed plate 21 and movable plate 23. Note that while Figure 1 illustrates only a single pair of fixed and movable plates, those skilled in the art would realize that clutch 20 could include multiple pairs of such plates.

A typical torque verses clutch position function is illustrated in Figure 2. Clutch torque/position curve 80 is initially zero for a range of engagements before initial touch point 81. Clutch torque rises monotonically with increasing clutch engagement. In the example illustrated in Figure 2, clutch torque rises slowly at first and then more steeply until the maximum clutch torque is reached upon full engagement at point 82. The typical clutch design calls for the maximum clutch torque upon full engagement to be about 1.5 times the maximum engine torque. This ensures that clutch 20 can transfer the maximum torque produced by engine 10 without slipping.

Clutch actuator 27 is coupled to movable plate 23 for control of clutch 20 from disengagement through partial engagement to full engagement. Clutch actuator 27 may be an electrical, hydraulic or pneumatic actuator and may be position or pressure controlled. Clutch actuator 27 controls the degree of clutch engagement according to a clutch engagement signal from clutch actuation controller 60.

Transmission input speed sensor 31 senses the rotational velocity of input shaft 25, which is the input to transmission 30. Transmission 30 provides selectable drive ratios to drive shaft 35 under the control of transmission shift controller 33. Drive shaft 35 is coupled to differential 40. Transmission output speed sensor 37 senses the rotational velocity of drive shaft 35. Transmission input speed sensor 31 and transmission output speed sensor 37 are preferably constructed in the same manner as engine speed sensor 13. In the preferred embodiment of the present invention, in which the motor vehicle is a large truck, differential 40 drives four axle shafts 41 to 44 that are in turn coupled to respective wheels 51 to 54.

Transmission shift controller 33 receives input signals from throttle 11, engine speed sensor 13, transmission input speed sensor 31 and transmission output speed sensor 37. Transmission shift controller 33 generates gear select signals for control of transmission 30 and clutch engage/disengage signals coupled to clutch actuation controller 60. Transmission shift controller 33 preferably changes the final gear ratio provided by transmission 30 corresponding to the throttle setting, engine speed, transmission input speed and transmission output speed. Transmission shift controller 33 provides respective engage and disengage signals to clutch actuation controller 60 depending on whether friction clutch 20 should be engaged or disengaged. Transmission shift controller also transmits a gear signal to clutch actuation controller 60. This gear signal permits recall of the set of coefficients corresponding to the selected gear.

Note transmission shift controller 33 forms no part of the present invention and will not be further described.

Clutch actuation controller 60 provides a clutch engagement signal to clutch actuator 27 for controlling the position of movable plate 23. This controls the amount of torque transferred by clutch 20 according to clutch torque/position curve 80 of Figure 2. Clutch actuation controller 60 operates under the control of transmission shift controller 33. Clutch actuation controller 60 controls the movement of moving plate 23 from disengagement to at least partial engagement or full engagement upon receipt of the engage signal from transmission shift controller 33. In the preferred embodiment it is contemplated that the clutch engagement signal will indicate a desired clutch position. Clutch actuator 27 preferably includes a closed loop control system controlling movable plate 23 to this desired position. It is also feasible for the clutch engagement signal to represent a desired clutch pressure with clutch actuator 27 providing closed loop control to this desired pressure. Depending on the particular vehicle, it may be feasible for clutch actuator 27 to operate in an open loop fashion. The exact details of clutch actuator 27 are not crucial to this invention and will not be further discussed.

Clutch actuation controller 60 preferably generates a predetermined open loop clutch disengagement signal for a ramped out disengagement of clutch 20 upon receipt of the disengage signal from transmission shift controller 33. No adverse oscillatory responses are anticipated for this predetermined open loop disengagement of clutch 20.

Figures 3 and 4 illustrate the two cases of starting the vehicle from a full stop. Figures 3 and 4 illustrate the engine speed and the transmission input speed during ideal clutch engagement. Figure 3 illustrates the case of launch. Figure 4 illustrates the case of creep.

Figure 3 illustrates the case of launch, that is starting out from a stop in order to proceed at a reasonable speed. Initially, the engine speed 90 is at idle. Thereafter engine speed 90 monotonically increases within the time frame of Figure 3. Engine speed 90 either increases or remains the same. Ideally engine speed 90 increases until the torque produced by engine 10 matches the torque required to accelerate the vehicle. At high load this engine speed may be in the mid range between the idle speed and the maximum engine speed. This constant engine speed corresponds to the engine torque required to match clutch torque and driveline torque and achieve a balance between engine output torque and the vehicle load torque. This torque level is the ideal clutch torque because a higher clutch torque would stall engine 10 and a lower clutch torque would allow the engine speed to increase too much. Ultimately the vehicle would accelerate to a speed where clutch 20 can be fully engaged. Thereafter the balance between engine torque and load torque is under the control of the driver via the throttle setting and clutch actuation controller 60 would continue to command full clutch engagement.

When the vehicle is stopped and clutch 20 fully disengaged, transmission input speed 100 is initially zero. This is the case for starting the vehicle. However, as further explained below, this same technique can be used for smooth clutch engagement upon shifting gears while moving. Thus the transmission input speed may initially be a value corresponding to the vehicle speed. Upon partial engagement of clutch 20, transmission input speed 100 increases and approaches engine speed 90 asymptotically. At a point 101, transmission input speed 100 is sufficiently close to engine speed 90 to achieve full engagement of clutch 20 without exciting the torsional compliance of the driveline of the vehicle. At this point clutch 20 is fully engaged. Thereafter transmission input speed 100 tracks engine speed 90 until clutch 20 is disengaged when the next higher final gear ratio is selected by transmission controller 33. The system preferably also operates for the case in which the vehicle is not stopped and the initial transmission input speed is nonzero.

Figure 4 illustrates the engine speed and transmission input speed for the case of creep. In the creep mode, clutch 20 must be deliberately slipped in order to match the available engine torque at an engine speed above idle and the required torque. Figure 4 illustrates engine speed 95 rising from idle to a plateau level. In a similar fashion input speed 105 rises from zero to a predetermined level. This predetermined level is less than the engine idle speed in this example. The creep mode is required when the desired vehicle speed implies a transmission input speed less than idle for the lowest gear ratio. The creep mode may also be required when the desired vehicle speed implies a transmission input speed above engine idle and engine 10 cannot produce the required torque at this engine speed. Note that there is a speed difference 107 between the engine speed 95 and the input speed 105 under quiescent conditions. This difference 107 represents the slip speed required for this creep operation.

Figure 5 illustrates schematically the control function of clutch actuation controller 60. As also illustrated in Figure 1, clutch actuation controller 60 receives the throttle signal from throttle 11, the engine speed signal from engine speed sensor 13 and the transmission input speed signal from transmission input speed sensor 31. Clutch actuation controller 60 illustrated in Figure 5 generates a clutch engagement signal that is supplied to clutch actuator 27 for operation of the friction clutch 20. Although not shown in Figure 5, the degree of clutch actuation, together with the throttle setting, the engine speed and the vehicle characteristics determine the transmission input speed that is sensed by transmission input speed sensor 31 and supplied to clutch actuation controller 60. Therefore, the control schematic illustrated in Figure 5 is a closed loop system.

The control function illustrated in Figure 5 is needed only for clutch positions between touch point 81 and full engagement. Clutch engagement less than that corresponding to touch point 81 provide no possibility of torque transfer because clutch 20 is fully disengaged. Clutch actuation controller 60 preferably includes some manner of detection of

the clutch position corresponding to touch point 81. Techniques for this determination are known in the art. As an example only, the clutch position at touch point 81 can be determined by placing transmission 30 in neutral and advancing clutch 20 toward engagement until transmission input speed sensor 31 first detects rotation. Upon receipt of the engage signal from transmission shift controller 33, clutch actuation controller 60 preferably rapidly advances clutch 20 to a point corresponding to touch point 81. This sets the zero of the clutch engagement control at touch point 81. Thereafter the clutch engagement is controlled by the control function illustrated in Figure 5.

Clutch actuation controller 60 is preferably realized via a microcontroller circuit. Inputs corresponding to the engine speed, the transmission input speed and the throttle setting must be in digital form. These input signals are preferably sampled at a rate consistent with the rate of operation of the microcontroller and fast enough to provide the desired control. As previously described, the engine speed, transmission input speed and transmission output speed are preferably detected via multitooth wheels whose teeth rotation is detected by magnetic sensors. The pulse trains detected by the magnetic sensors are counted during predetermined intervals. The respective counts are directly proportional to the measured speed. For proper control the sign of the transmission input speed signal must be negative if the vehicle is moving backwards. Some manner of detecting the direction of rotation of input shaft 25 is needed. Such direction sensing is conventional and will not be further described. The throttle setting is preferably detected via an analog sensor such as a potentiometer. This analog throttle signal is digitized via an analog-to-digital converter for use by the microcontroller. The microcontroller executes the processes illustrated in Figures 5 by discrete difference equations in a manner known in the art. The control processes illustrated in Figure 5 should therefore be regarded as an indication of how to program the microcontroller embodying the invention rather than discrete hardware. It is feasible for the same microcontroller, if of sufficient capacity and properly programmed, to act as both clutch actuation controller 60 and as transmission shift controller 33. It is believed that an Intel 80C196 microcontroller has sufficient computation capacity to serve in this manner.

The throttle signal received from throttle 11 is supplied to launch/creep selector 61 and to creep speed reference 62. Launch/creep selector 61 determines from the throttle signal whether to operate in the launch mode or to operate in the creep mode. In the preferred embodiment of the present invention, launch/creep selector 61 selects the launch mode if the throttle signal indicates greater than 25% of the full throttle setting. In other cases launch/creep selector 61 selects the creep mode.

Creep speed reference 62 receives the throttle signal and the engine speed signal and generates a creep speed reference signal. This creep speed reference signal is determined as follows:

$$R_{crp} = E_{sp} \frac{T}{T_{ref}} \qquad (1)$$

where: R_{crp} is the creep speed reference signal; E_{sp} is the measured engine speed; T is the throttle signal; and T_{ref} is a throttle reference constant equal to the throttle signal for 25% full throttle. The creep speed reference signal is the product of the engine speed signal and the ratio of the actual throttle to 25% full throttle. No creep speed reference signal is required for throttle settings above 25% of full throttle because the launch mode is applicable rather than the creep mode. Note that this creep speed reference signal makes the speed reference signal continuous even when switching between the launch mode and the creep mode. Thus no instabilities are induced if changes in the throttle setting causes switching between the two modes.

Mode select switch 63 determines the mode of operation of clutch actuation controller 60. Mode select switch 63 receives the mode selection determination made by launch/creep selector 61. Mode select switch 63 selects either the engine speed signal or the creep speed reference signal depending upon the mode determined by launch/creep selector 61. In the event that the launch mode is selected mode select switch 63 selects the engine speed for control. Thus in the launch mode the clutch engagement is controlled so that the transmission input speed matches the engine speed. In the event that the creep mode is selected mode select switch 63 selects the creep speed reference signal for control. In creep mode the clutch engagement is controlled to match transmission input speed to the creep speed reference signal. This is equivalent to controlling clutch engagement to match the actual clutch slip to desired slip speed. In either mode, the speed reference signal is a transmission input speed reference.

As noted above, mode select switch 63 selects a speed reference signal for control. Clutch actuation controller 60 includes an integral function. The transmission input speed from transmission input speed sensor 31 is subtracted from the speed reference signal selected by mode select switch 63 in algebraic summer 64. Ignoring for the moment threshold detector 75, and switches 76 and 77, integrator 65 integrates this difference signal, which is the error between the desired transmission input speed from mode select switch 63 and the measured transmission input speed. The integrated difference signal is supplied to algebraic summer 67 and to a second integrator 66. Integrator 66 integrates the integral of the error, thus forming a second integral of this error. Algebraic summer 67 sums the speed reference signal from mode select switch 63, the integrated error from integrator 65 and the second integral of the error from integrator

66.

Algebraic summer 67 supplies the input to prefilter 68. Prefilter 68 is employed to shape the closed loop transient response of automatic clutch controller 60. This shaping of the transient response has the goal of achieving asymptotic approach of the input speed to the reference speed. The character of prefilter 68 and its manner of determination will be further described below.

The prefiltered signal from prefilter 68 is supplied to algebraic summer 69. Algebraic summer 69 also receives the measured transmission input speed signal from transmission input speed sensor 31. Algebraic summer 69 forms the difference between the prefiltered signal from prefilter 68 and the transmission input speed. This difference is supplied to compensator 70. Compensator 70 includes an approximate inverse model of the torsional oscillatory response of the vehicle to torque inputs. Compensator 70 includes a gain versus frequency function selected to reduce variations in the closed loop response of clutch actuation controller 60 due to variations in the transfer function of the vehicle driveline. Determination of the transfer function of compensator 70 will be further described below.

A feedforward signal is provided in the clutch engagement signal via an engine speed differential signal. The engine speed signal is suitably filtered via low pass filter 72 to reduce noise in the differential signal. Differentiator 73 forms a differential signal proportional to the rate of change in the engine speed. This engine speed differential signal and its integral formed by integrator 74 are supplied to algebraic summer 71. Algebraic summer 71 sums the output of compensator 70, the engine speed differential signal from differentiator 73 and the integral signal from integrator 74 to form the clutch engagement signal. Clutch actuator 27 employs this clutch engagement signal to control the degree of clutch engagement.

The feedforward signal permits better response of clutch actuation controller 60 when the engine speed is accelerating. Under conditions of engine speed acceleration the feedforward signal causes rapid engagement of clutch 20 proportional to the rate of engine acceleration. The engine speed can increase rapidly under full throttle conditions before the driveline torque is established. This is because the speed of response of clutch actuation controller 60 without this feedforward response is low compared with the peak engine speed of response. With this feedforward response rapid engine acceleration results in more rapid than otherwise clutch engagement. The additional clutch engagement tends to restrain increase in engine speed by requiring additional torque from the engine. When the engine speed reaches a constant value, the differential term decays to zero and integrator 74 retains the clutch engagement needed to restrain engine speed. Other portions of the control function then serve to provide asymptotic convergence of the transmission input speed to the reference speed.

Provision of the integral and double integral signals in the input to prefilter 68 serves to ensure clutch lockup when operating in the launch mode. The second integral ensures clutch lockup even if the engine speed in increasing. The integration rates of integrators 65 and 66 can be adjusted by corresponding integration coefficients k_{I1} and k_{I2}. The existence of any long term difference between the speed reference signal selected by mode select switch 63 and the transmission input speed generates an increasing integral signal. Any such integral signal serves to drive the clutch engagement signal toward full clutch engagement. This ensures that clutch 20 is fully engaged at point 101 at some predetermined maximum time following start up of the vehicle when in the launch mode. In the creep mode, integrators 65 and 66 ensure that there is no long term error between the creep speed reference signal and the transmission input speed.

The integral function and the second integral function are preferably disabled when the rate of engine speed increase falls below a predetermined threshold. This level could be zero, disabling the first and second integral functions when the engine speed decreases. Threshold detector 75 determines when to disable integrators 65 and 66 based on the differential signal. The rate of engine speed increase would typically fall below the threshold upon too rapid clutch engagement for the current engine speed and vehicle torque demand. Switches 76 and 77 are normally closed, enabling integrators 65 and 66. If the rate of change of engine speed formed by differentiator 73 is below the threshold of threshold detector 75, then threshold detector 75 trips. This opens switches 76 and 77, and disables further integration in integrators 65 and 66. The additional clutch advancement caused by integrators 65 and 66 ceases. In this case the clutch would hold for a time at a steady position. This permits a torque balance between the engine output torque and the vehicle load torque. This torque balance tends to keep engine 10 at a constant speed. This generally occurs under high vehicle load conditions when the vehicle takes longer to accelerate. The engine torque transmitted via clutch 20 to the vehicle load tends to accelerate the vehicle. Clutch lockup is delayed during the interval when integrators 65 and 66 are disabled. Clutch lockup may still occur under these conditions if the vehicle accelerates to a high enough speed so that the transmission input speed reaches the engine speed. When the vehicle load permits the rate of change of engine speed to again exceed the threshold, then integrators 65 and 66 are re-enabled. This permits integrators 65 and 66 to drive the clutch engagement signal to clutch lockup. Note that during the interval when integrators 65 and 66 are disabled and clutch lockup is delayed, the interval to clutch lockup can be shortened by increasing the throttle. This provides additional engine torque, permitting an engine speed increase and re-enabling the integrators.

This switching of integrators 65 and 66 provides adaptive clutch engagement. Clutch engagement is rapid under conditions of engine acceleration, which generally occurs only during light vehicle loads. Under conditions of high

vehicle loads, full clutch engagement is delayed to prevent engine stalling. Thus this technique complements the feed-forward technique that causes rapid clutch engagement when the engine is accelerating.

Prefilter 68 and compensator 70 perform differing and complementary functions in clutch actuation controller 60. The transfer functions of prefilter 68 and compensator 70 are determined as follows. The transfer function of compensator 70 is selected to reduce sensitivities of the closed loop transfer function to driveline parameter variations. This is achieved by providing sufficient loop gain as a function of frequency. If the sensitivity of the closed loop transfer function $H(\omega)$ with respect to the transfer function of the driveline $G(\omega)$ is $S^{H(\omega)}_{G(\omega)}$, then

$$S^{H(\omega)}_{G(\omega)} = \frac{1}{(1 + C(\omega)\,G(\omega))} \tag{2}$$

where $C(\omega)$ is the transfer function of compensator 70. Inspection of this relationship reveals that the sensitivity $S^{H(\omega)}_{G(\omega)}$ can be reduced arbitrarily to zero by increasing the compensator gain. There are practical limits to the maximum compensator gain because of stability and noise problems. Thus the transfer function $C(\omega)$ of compensator 70 is selected high enough at all frequencies ω to limit the variations in the closed loop transfer function to an acceptable level set as a design criteria.

Compensator 70 includes an approximate inverse model of the torsional oscillatory response. In the typical heavy truck to which this invention is applicable, the torsional compliance of the driveline causes the driveline transfer function to have a pair of lightly damped poles that may range from 2 to 5 Hz. The exact value depends upon the vehicle parameter values. The inverse response of compensator 70 provides a notch filter in the region of these poles. The frequency band of the notch is sufficiently broad to cover the range of expected vehicle frequency responses. This frequency band is preferably achieved employing two pairs of zeros whose frequencies are spread over the frequency range of the vehicle response. Thus compensator 70 provides plural complex zeros in the frequency range of these poles of the vehicle response to attenuate the oscillatory response. The typical heavy truck also includes a pair of complex zeros in the frequency range from 1 to 2 Hz. These complex zeros tend to reduce the system loop gain and hence cause the system to be more sensitive to variations in vehicle characteristics in this frequency range. Compensator 70 preferably provides a pair of complex poles in this frequency range to increase the loop gain and reduce sensitivity to variations in vehicle characteristics. Thus the total response of the closed loop system has highly damped eigen values providing a less oscillatory system.

Prefilter 68 is employed to reliably achieve a desired closed loop transient response. The transfer function $H(\omega)$ of the closed loop system without prefilter 68 is:

$$H(\omega) = \frac{C(\omega)\,G(\omega)}{(1 + C(\omega)\,G(\omega))} \tag{3}$$

where $C(\omega)$ is the transfer function of compensator 70 and $G(\omega)$ is the transfer function of the driveline. The above noted design for compensator 70 takes into account only reduction in sensitivity to variations in the driveline response $G(\omega)$. This typically results in a closed loop response $H(\omega)$ having an inappropriate time response. The design goal it to actuate clutch 20 to achieve asymptotic convergence of the transmission input speed to engine speed. The transfer function $H(\omega)$ with prefilter 68 is:

$$H(\omega) = \frac{F(\omega)\,C(\omega)\,G(\omega)}{(1 + C(\omega)\,G(\omega))} \tag{4}$$

where $F(\omega)$ is the transfer function of prefilter 68. Prefilter 68 is a low pass filter with the pass band related to the design rate of asymptotic convergence.

The above outlined determination of the response character of prefilter 68 and compensator 70 corresponds to the quantitative feedback theory of Horowitz. This theory is exemplified in "Quantative Feedback Theory" by I. M. Horowitz, IEE Proceedings, Vol. 129, PT.d, no. 6, November 1982. This selection of the response of prefilter 68 and compensator 70 results in a system that is robust, that is, capable of properly responding to widely varying vehicle conditions.

As noted above, the elements of Figure 5 are preferably implemented via discrete difference equations in a microcontroller. In the preferred embodiment the i-th value of the output P_i of prefilter 68 is given by:

$$P_i = k_{P1}\, I_{i-1} + k_{P2}\, I_i + k_{P3}\, P_{i-1} + k_{P4}\, P_{i-2} \tag{5}$$

where; I_i is the current value of the prefilter input; I_{i-1} is the immediately preceding value of the prefilter input; P_{i-1} is the immediately preceding value of the prefilter output; P_{i-2} is the next preceding value of the prefilter output; and where the k_{P_n} are coefficients with $k_{P1} = 0.00015$, $k_{P2} = 0.00015$, $k_{P3} = 1.9677$, and $k_{P4} = -0.9860$.

The discrete difference equation of compensator 70 is preferably implemented in three stages. This enables the compensator coefficients to have sufficiently fewer significant figures for a 16 bit integer digital implementation of this process. The i-th value of the first intermediate variable $F1_i$ is given by:

$$F1_i = k_{C1}\, C_i + k_{C2}\, C_{i-1} + k_{C3}\, C_{i-2} + k_{C4}\, F1_{i-1} + k_{C5}\, F1_{i-2} \qquad (6)$$

where: C_i is the current value of the compensator input; C_{i-1} is the immediately preceding value of the compensator input; C_{i-2} is the next preceding value of the compensator input; $F1_{i-1}$ is the immediately preceding value of the first intermediate variable; $F1_{i-2}$ is the next preceding value of the first intermediate variable; and where the k_{C_n} are coefficients with $k_{C1} = 0.667$, $k_{C2} = = -1.16$, $k_{C3} = 0.5532$, $k_{C4} = 1.482$, and $k_{C5} = -0.5435$. Note that the successive compensator input values C_i are computed from successive differences between the prefilter output and the transmission input speed. The i-th value of the second intermediate variable $F2_i$ is given by:

$$F2_i = k_{C6}\, F1_i + k_{C7}\, F1_{i-1} + k_{C8}\, F1_{i-2} + k_{C9}\, F2_{i-1} + k_{C10}\, F2_{i-2} \qquad (7)$$

where: $F1_i$ is the current value of the first intermediate variable; $F1_{i-1}$ is the immediately preceding value of the first intermediate variable; $F1_{i-2}$ is the next preceding value of the first intermediate variable; $F2_{i-1}$ is the immediately preceding value of the second intermediate variable; $F2_{i-2}$ is the next preceding value of the second intermediate variable; and where the k_{C_n} are coefficients with $k_{C6} = 0.2098$, $k_{C7} = -0.39$, $k_{C8} = 0.189$, $k_{C9} = 1.8432$, and $k_{C10} = -0.8518$. Lastly, the i-th value of the compensator output O_i is:

$$O_i = k_{C11}\, F2_i + k_{C12}\, F2_{i-1} + k_{C13}\, O_{i-1} \qquad (8)$$

where: $F2_i$ is the current value of the second intermediate variable; $F2_{i-1}$ is the immediately preceding value of the second intermediate variable; O_{i-1} is the immediately preceding value of the compensator output; and where the k_{C_n} are coefficients with $k_{C11} = 0.25$, $k_{C112} = -0.245$, and $k_{C13} = 0.995$.

The present invention can be advantageously employed for clutch re-engagement following shifts of the transmission. In this event the same control processes illustrated in Figure 5 would be employed, including the above listed discrete difference equations for prefilter 68 and compensator 70. The control processes for transmission shifts would differ from the preceding description in selection of the coefficients k_{P1} to k_{P4} and k_{C1} to k_{C13}. A particular set of these coefficients k_n would be recalled from coefficient memory 75 depending upon the gear signal from transmission shift controller 33. The selected set of coefficients may also include coefficients of integration for integrators 65, 66 and 74, and coefficients for filter 69 and differentiator 70. In other respects the invention would operate the same as described above.

The control processes of the present invention are robust with regard to variations in vehicle response. It is believed that the automatic clutch controller herein described is capable of handling changes in vehicle loading within a single vehicle and variations in response between differing combinations of engine, clutch and driveline oscillatory response between different vehicles. Thus the automatic clutch controller of this invention need not be particularized for a particular vehicle. Thus the invention automatic clutch controller is easier to manufacture for a variety of vehicles.

Claims

1. An automatic clutch controller for a friction clutch (20) having an input shaft (15) connected to an engine (10) and an output connected to a input shaft (25), and at least one inertially-loaded traction wheel (51) connected to the input shaft having a torsional compliance exhibiting an oscillatory response to torque inputs, the controller comprising an engine speed sensor (13) connected to the engine for generating an engine speed signal corresponding to the rotational speed of the engine, a transmission input speed sensor (31) connected to the input shaft for generating a transmission input speed signal corresponding to the rotational speed of the input shaft, a clutch actuator (27) connected to the friction clutch for controlling engagement of the friction clutch from disengaged to fully engaged according to a clutch engagement signal, and a clutch actuator controlling unit (64-78), wherein the

controller is characterized by a reference speed generator (61-63) coupled to the engine speed signal for generating a reference speed signal, and the controlling unit is connected to the reference speed generator, the transmission input speed sensor and the clutch actuator and includes a prefilter (68) connected to the reference speed generator for generating a filtered reference speed signal, a first algebraic summer (69) connected to the transmission input speed sensor and the prefilter generating a first algebraic sum signal corresponding to the difference between (1) the filtered reference speed signal and (2) the transmission input speed signal, and a compensator (70) connected to the first algebraic summer for generating the clutch engagement signal for supply to the clutch actuator for engaging the friction clutch in a manner causing the transmission input speed signal to asymptotically approach the reference speed signal.

2. The automatic clutch controller as claimed in claim 1, wherein: the compensator has a transfer function having a notch filter with a frequency band in the range of the expected frequency of the oscillatory response to torque inputs of the at least one inertially-loaded traction wheel, the compensator thereby reducing the oscillatory response to torque inputs of the at least one inertially-loaded traction wheel.

3. The automatic clutch controller as claimed in claim 1, wherein: the compensator has a transfer function having a region of increased gain in the frequency range where the expected response to torque inputs of the at least one inertially-loaded traction wheel is a minimum, the compensator thereby increasing loop gain and maintaining reduced sensitivity of the controller to variations in the response to torque inputs of the at least one inertially-loaded traction wheel.

4. The automatic clutch controller as claimed in claim 1 wherein: the prefilter is a low pass filter having a cutoff frequency selected to provide a desired transient response of the transmission input speed signal to a step function in the reference speed signal.

5. The automatic clutch controller as claimed in claim 1, wherein the controlling unit further includes:

a second algebraic summer 64 connected to the reference speed generator and the transmission input speed sensor for forming a second algebraic sum signal corresponding to the difference between (1) the reference speed signal and (2) the transmission input speed signal, a first integrator 65 connected to the second algebraic summer for forming a first integral signal corresponding to the time integral of the second algebraic sum signal, and a third algebraic summer 67 connected to the reference speed generator and the first integrator for forming a third algebraic sum signal corresponding to the sum of (1) the reference speed signal and (2) the first integral signal, the third algebraic sum signal supplied to the prefilter whereby the prefilter is connected to the reference speed generator via the third algebraic summer.

6. The automatic clutch controller as claimed in claim 5, wherein the controlling unit further includes:

a second integrator 66 connected to the first integrator for forming a second integral signal corresponding to the time integral of the first integral signal, and the third algebraic summer is further connected to the second integrator for forming the third algebraic sum signal corresponding to the sum of (1) the reference speed signal, (2) the first integral signal and (3) the second integral signal.

7. The automatic clutch controller as claimed in claim 1, wherein the controlling unit further includes:

a differentiator 73 connected to the engine speed sensor for generating a differential signal corresponding to the rate of change of the engine speed signal, and a fourth algebraic summer 71 connected to the compensator and the differentiator for generating the clutch actuation signal corresponding to the sum of (1) the output of the compensator and (2) the differential signal.

8. The automatic clutch controller as claimed in claim 7, wherein the controlling unit further includes a low pass filter 72 disposed between the engine speed sensor and the differentiator.

9. The automatic clutch controller as claimed in claim 7, wherein the controlling unit further includes a third integrator 74 connected to the differentiator for forming a third integral signal corresponding to the time integral of the differential signal, and the fourth algebraic summer is further connected to the third integrator and generates the clutch actuation signal corresponding to the sum of (1) the output of the compensator, (2) the differential signal and (3) the third integral signal.

10. The automatic clutch controller as claimed in claim 9, wherein the controlling unit further includes:

a second algebraic summer 64 connected to the reference speed generator and the transmission input speed

sensor for forming a second algebraic sum signal corresponding to the difference between (1) the reference speed signal and (2) the transmission input speed signal, a threshold detector 75 connected to the differentiator for determining whether the differential signal is less than a predetermined threshold, a first switch 76 connected to the second algebraic summer and the threshold detector for generating a first switch output (1) equal to zero when the threshold detector determines the differential signal is less than the predetermined threshold, and (2) otherwise corresponding to the second algebraic sum signal, a first integrator connected to the first switch for forming a first integral signal corresponding to the time integral of the first switch output, and a third algebraic summer connected to the reference speed generator and the first integrator for forming a third algebraic sum signal corresponding to the sum of (1) the reference speed signal and (2) the first integral signal, the third algebraic sum signal supplied to the prefilter whereby the prefilter is connected to the reference speed generator via the third algebraic summer.

11. The automatic clutch controller as claimed in claim 10, wherein the predetermined threshold of the threshold detector is zero.

12. The automatic clutch controller as claimed in claim 10, wherein the controlling unit further includes:

a second switch 77 connected to the threshold detector and the first integrator for generating a second switch output (1) equal to zero when the threshold detector determines the differential signal is less than the predetermined threshold, and (2) otherwise corresponding to the first integral signal, a second integrator connected to the second switch for forming a second integral signal corresponding to the time integral of the second switch output, and the third algebraic summer is further connected to the second integrator for forming the third algebraic sum signal corresponding to the sum of (1) the reference speed signal, (2) the first integral signal and (3) the second integral signal.

13. The automatic clutch controller as claimed in claim 1 wherein: the reference speed generator is connected to the engine speed sensor and generates the reference speed signal corresponding to the engine speed signal; and the compensator generates the clutch engagement signal for fully engaging the friction clutch within a predetermined interval of time after initial partial engagement.

14. The automatic clutch controller as claimed in claim 1, wherein: the compensator generates the clutch engagement signal indicative of desired clutch position; and the clutch actuator controls the position of the friction clutch corresponding to the desired clutch position indicated by the clutch engagement signal.

15. The automatic clutch controller as claimed in claim 1, wherein: the compensator generates the clutch engagement signal indicative of desired clutch pressure; and the clutch actuator controls the pressure of the friction clutch corresponding to the desired clutch pressure indicated by the clutch engagement signal.

16. The automatic clutch controller as claimed in claim 1, the automatic clutch controller further including a throttle for control of torque generated by the engine and a throttle sensor connected to the throttle for generating a throttle signal indicative of throttle position, the controlling unit further comprising:

the reference speed generator being connected to the engine speed sensor and the throttle sensor for generating the reference speed signal corresponding to the engine speed signal and the throttle signal.

17. The automatic clutch controller as claimed in claim 16, wherein:

the reference speed generator generates the reference speed signal as follows

$$S_{ref} = E_{sp} \frac{T}{T_{ref}}$$

where: S_{ref} is the reference speed signal; E_{sp} is the engine speed signal; T is the throttle signal; and T_{ref} is a throttle reference constant equal to the throttle signal for a predetermined throttle position.

18. The automatic clutch controller as claimed in claim 1, the automatic clutch controller further including a throttle for control of torque generated by the engine and a throttle sensor connected to the throttle for generating a throttle signal indicative of throttle position, the controlling unit comprising:

the reference speed generator being further connected to the throttle and including a launch/creep selector 61 connected to the throttle sensor for selecting either a launch mode or a creep mode based upon the magnitude of the throttle signal, a creep speed reference generator 62 connected to the engine speed sensor and the throttle

sensor for generating a creep speed reference signal corresponding to the engine speed signal and the throttle signal, and a mode selection switch 63 connected to the engine speed sensor, the launch/creep selector and the creep speed reference generator for selectively generating a reference speed signal corresponding to (1) the engine speed signal if the launch mode is selected and (2) the creep speed reference signal if the creep mode is selected.

19. The automatic clutch controller as claimed in claim 18, wherein:
the launch/creep selection selects the launch mode if the throttle signal indicates a throttle position of greater than a predetermined throttle position and otherwise selects the creep mode.

20. The automatic clutch controller as claimed in claim 19, wherein:
the predetermined throttle position of the launch/creep selector is 25% of full throttle.

21. The automatic clutch controller as claimed in claim 18, wherein:
the creep speed reference generator generates the creep speed reference signal as follows

$$S_{ref} = E_{sp} \frac{T}{T_{ref}}$$

where: S_{ref} is the creep speed reference signal; E_{sp} is the engine speed signal; T is the throttle signal; and T_{ref} is a throttle reference constant equal to the throttle signal for the predetermined throttle position.

22. The automatic clutch controller as claimed in one of claims 1 to 21, wherein the controlling unit is implemented by discrete difference equations executed by a microcontroller and includes:
the prefilter and the compensator being implemented in discrete difference equations.

23. The automatic clutch controller as claimed in one of claims 1 to 21, wherein a transmission having an input shaft connected to the output shaft of the friction clutch provides a selectable gear ratio to the output shaft and a transmission shift controller connected to the transmission for controlling the gear ratio selected by the transmission, wherein:
the controlling unit is implemented via discrete difference equations executed by a microcontroller and including a coefficient memory for storing a plurality of sets of coefficients, one set of coefficients corresponding to each selectable gear ratio of the transmission, the prefilter is implemented in discrete difference equations employing a set of coefficients recalled from said coefficient memory corresponding to the gear ratio of the transmission, and the compensator is implemented in discrete difference equations employing a set of coefficients recalled from said coefficient memory corresponding to the gear ratio of the transmission.

24. A method of automatically controlling a friction clutch, the friction clutch being controlled by a controller and the friction clutch (20) having an input shaft (15) connected to an engine (10) and an output connected to an input shaft (25), and at least one inertially-loaded traction wheel (51) connected to the input shaft having a torsional compliance exhibiting an oscillatory response to torque inputs, the controller comprising an engine speed sensor (13) connected to the engine for generating an engine speed signal corresponding to the rotational speed of the engine, a transmission input speed sensor (31) connected to the input shaft for generating a transmission input speed signal corresponding to the rotational speed of the input shaft, a clutch actuator (27) connected to the friction clutch for controlling engagement of the friction clutch from disengaged to fully engaged according to a clutch engagement signal, and a clutch actuator controlling unit (64-78), wherein the method is characterized by generating a reference speed signal by means of a reference speed generator (61-63) coupled to the engine speed signal, and the controlling unit being connected to the reference speed generator, the transmission input speed sensor and the clutch actuator and generating a filtered reference speed signal by means of a prefilter (68) connected to the reference speed generator summing algebraically, by means of a first algebraic summer (69), the prefiltered reference speed signal and the transmission input speed signal, outputting the result to a compensator (70), said compensator outputting a compensated signal, inputting said compensated signal to a second algebraic summer (71) the second algebraic summer (71) summing input signals from differential engine speed signals from differentiator (73), and integrated engine speed signals from integrator (74) to give a clutch engagement signal to control the friction clutch engagement in a manner which causes the transmission input speed signal to asymptotically approach the reference speed signal.

25. A method according to Claim 24, wherein:
the oscillatory response to torque inputs of the at least one inertially-loaded traction wheel is reduced by means of a compensator, the compensator includes transfer function having a notch filter with a frequency band in the range of the expected frequency of the oscillatory response to torque inputs of the at least one inertially-loaded traction wheel.

26. A method according to Claim 24, wherein:
the compensator increases loop gain and maintains reduced sensitivity of the controller to variations in the response to torque inputs of the at least one inertially-loaded traction wheel by means of a transfer function having a region of increased gain in the frequency range where the expected response to torque inputs of the at least one inertially-loaded traction wheel is a minimum.

Patentansprüche

1. Automatische Kupplungssteuerung für eine Reibkupplung (20) mit einer Eingangswelle (15), die mit einem Motor (10) verbunden ist, und mit einem Ausgang, der mit einer Eingangswelle (25) verbunden ist, wobei wenigstens ein schwungbelastetes Antriebsrad (51) mit der Eingangswelle verbunden ist, die ein Drehverhalten aufweist, das für Drehmomenteingangssignale eine Schwingantwort zeigt, wobei zu der Steuerung ein Motordrehzahlsensor (13), der zur Erzeugung eines Motordrehzahlsignals, das der Drehzahl des Motors entspricht, mit dem Motor verbunden ist, ein Getriebeeingangswellendrehzahlsensor (31), der mit der Eingangswelle verbunden ist, um ein Getriebeeingangsdrehzahlsignal zu erzeugen, das der Drehzahl der Eingangswelle entspricht, ein Kupplungsaktuator (27), der mit der Reibungskupplung verbunden ist, um das Einkuppeln der Reibungskupplung aus dem ausgekuppelten in den vollständig eingekuppelten Zustand gemäß eines Kupplungseinrücksignals zu kontrollieren, und eine Steuereinheit (64-78) des Kupplungsaktuators gehören, wobei die Steuerung durch einen Referenzdrehzahlgenerator (61-63) gekennzeichnet ist, der an das Motordrehzahlsignal angeschlossen ist, um ein Referenzdrehzahlsignal zu erzeugen, wobei die Steuereinheit mit dem Referenzdrehzahlgenerator, dem Getriebeeingangswellendrehzahlsensor und dem Kupplungsaktuator verbunden ist und ein Vorfilter (68) aufweist, das mit dem Referenzdrehzahlgenerator verbunden ist, um ein gefiltertes Referenzdrehzahlsignal zu erzeugen, ein erster algebraischer Summierer (69) ist mit dem Getriebeeingangswellendrehzahlsensor und dem Vorfilter verbunden und erzeugt ein erstes algebraisches Summensignal, das der Differenz zwischen (1), dem gefilterten Referenzdrehzahlsignal und (2) dem Getriebeeingangswellendrehzahlsignal entspricht, und ein Kompensator (70) ist mit dem ersten algebraischen Summierer verbunden, um das Kupplungseinrücksignal zu erzeugen, das an den Kupplungsaktuator geliefert wird, um die Reibkupplung in einer Weise einzurücken, die das Getriebeeingangswellendrehzahlsignal veranlasst, sich asymptotisch an das Referenzdrehzahlsignal anzunähern.

2. Automatische Kupplungssteuerung nach Anspruch 1, bei der:
der Kompensator eine Übertragungsfunktion aufweist, die ein Sperrfilter mit einem Frequenzband in dem Bereich der Frequenz aufweist, die für die Schwingantwort für Drehmomenteingangssignale des wenigstens einen schwungbelasteten Antriebsrads erwartet wird, so dass der Kompensator die Schwingreaktion auf Drehmomentsignale des wenigstens einen schwungbelasteten Antriebsrads reduziert.

3. Automatische Kupplungssteuerung nach Anspruch 1, bei der:
der Kompensator eine Übertragungsfunktion aufweist, die wenigstens einen Bereich erhöhter Verstärkung in dem Frequenzbereich hat, in dem die erwartete Reaktion auf Drehmomenteingangssignale des wenigstens einen schwungbelasteten Antriebsrads minimal ist, so dass der Kompensator die Schleifenverstärkung erhöht und eine verminderte Empfindlichkeit der Steuerung auf Veränderungen in der Reaktion auf Drehmomenteingangssignale des wenigstens einen schwungbelasteten Antriebsrads beibehält.

4. Automatische Kupplungssteuerung nach Anspruch 1, bei der:
das Vorfilter ein Tiefpassfilter mit einer Eckfrequenz ist, die so ausgewählt ist, dass sie eine gewünschte Übergangsantwort des Getriebeeingangswellendrehzahlsignals auf eine Sprungfunktion des Referenzdrehzahlsignals liefert.

5. Automatische Kupplungssteuerung nach Anspruch 1, bei der die steuernde Einheit außerdem aufweist:
einen zweiten algebraischen Summierer (64), der mit dem Referenzdrehzahlgenerator und dem Getriebeeingangswellendrehzahlsensor verbunden ist, um ein zweites algebraisches Summensignal zu erzeugen, das der Differenz zwischen (1), dem Referenzdrehzahlsignal und (2) dem Getriebeeingangswellendrehzahlsignal ent-

spricht, wobei ein erster Integrator (65) mit dem zweiten algebraischen Summierer verbunden ist, um ein erstes Integralsignal zu erzeugen, das dem Zeitintegral des zweiten algebraischen Summensignals entspricht, wobei ein dritter algebraischer Summierer (67) mit dem Referenzdrehzahlgenerator und dem ersten Integrator verbunden ist, um ein drittes algebraisches Summensignal zu erzeugen, das der Summe aus (1) dem Referenzdrehzahlsignal und (2) dem ersten Integralsignal entspricht, wobei das dritte algebraische Summensignal zu dem Vorfilter geliefert wird, so dass das Vorfilter über den dritten algebraischen Summierer mit dem Referenzdrehzahlgenerator verbunden ist.

6. Automatisches Kupplungssteuerung nach Anspruch 5, wobei die Steuereinheit außerdem aufweist:
einen zweiten Integrator (66), der mit dem ersten Integrator verbunden ist, um ein zweites Integralsignal zu bilden, das dem Zeitintegral des ersten Integralsignals entspricht, wobei der dritte algebraische Summierer außerdem mit dem zweiten Integrator verbunden ist, um das dritte algebraische Summensignal zu bilden, das der Summe aus (1) dem Referenzdrehzahlsignal (2) dem ersten Integralsignal und (3) dem zweiten Integralsignal entspricht.

7. Automatische Kupplungssteuerung nach Anspruch 1, bei der die Steuereinheit außerdem aufweist:
einen Differenzierer (73), der mit dem Motordrehzahlsensor verbunden ist, um ein Differenzsignal zu erzeugen, das der Änderungsgeschwindigkeit des Motordrehzahlsignals entspricht und wobei ein vierter algebraischer Summierer (71) mit dem Kondensator und dem Differenziator verbunden ist, um das Kupplungsbetätigungssignal zu erzeugen, das der Summe aus (1) dem Ausgangssignal des Kompensators und (2) dem Differenzialsignal entspricht.

8. Automatische Kupplungssteuerung nach Anspruch 7, bei der die Steuereinheit außerdem ein zwischen dem Motordrehzahlsensor und dem Differentiator angeordnetes Tiefpassfilter (72) aufweist.

9. Automatische Kupplungssteuerung nach Anspruch 7, bei der die Steuereinheit außerdem einen dritten Integrator (74) aufweist, der mit dem Differentiator verbunden ist, um ein drittes Integralsignal zu bilden, das dem Zeitintegral des Differenzialsignals entspricht, wobei der vierte algebraische Summierer außerdem mit dem dritten Integrator verbunden ist und das Kupplungsbetätigungssignal erzeugt, das der Summe aus (1) dem Ausgangssignal des Kompensators, (2) dem Differenzialsignal und (3) dem dritten Integralsignal entspricht.

10. Automatische Kupplungssteuerung nach Anspruch 9, bei der die Steuereinheit außerdem aufweist:
einen zweiten algebraischen Summierer (64), der mit dem Referenzdrehzahlgenerator und dem Getriebeeingangswellendrehzahlsensor verbunden ist, um ein zweites algebraisches Summensignal zu erzeugen, das der Differenz zwischen (1) dem Referenzdrehzahlsignal und (2) dem Getriebeeingangswellendrehzahlsignal entspricht, wobei mit dem Differentiator ein Schwellwertdetektor (75) verbunden ist, um zu bestimmen, ob das Differentialsignal geringer ist als ein vorbestimmter Schwellwert, wobei ein erster Schalter (76) mit dem zweiten algebraischen Addierer und dem Schwellwertdetektor verbunden ist, um ein erstes Schalterausgangssignal zu erzeugen, das (1) gleich Null ist, wenn der Schwellwertdetektor bestimmt, dass das Differenzialsignal geringer ist als die vorbestimmte Schwelle und das (2) ansonsten dem zweiten algebraischen Summensignal entspricht, wobei ein erster Integrator mit dem ersten Schalter verbunden ist, um ein erstes Integralsignal zu erzeugen, das dem Zeitintegral des ersten Schalterausgangssignals entspricht und wobei ein dritter algebraischer Summierer mit dem Referenzdrehzahlgenerator und dem ersten Integrator verbunden ist, um ein drittes algebraisches Summensignal zu erzeugen, das der Summe aus (1) dem Referenzdrehzahlsignal und (2) dem ersten Integralsignal entspricht, wobei das dritte algebraische Summensignal zu dem Vorfilter geliefert wird, so dass das Vorfilter über den dritten algebraischen Summierer mit dem Referenzdrehzahlsignalgenerator verbunden ist.

11. Automatische Kupplungssteuerung nach Anspruch 10, bei der die vorbestimmte Schwelle des Schwellwertgenerators Null ist.

12. Automatische Kupplungssteuerung nach Anspruch 10, bei der die Steuereinheit außerdem aufweist:
einen zweiten Schalter (77), der mit dem Schwellwertdetektor und dem ersten Integrator verbunden ist, um ein zweites Schalterausgangssignal zu erzeugen, das (1) gleich Null ist, wenn der Schwellwertdetektor bestimmt, dass das Differenzialsignal kleiner ist als der festgelegte Schwellwert, und das (2) ansonsten dem ersten Integralsignal entspricht, wobei ein zweiter Integrator mit dem zweiten Schalter verbunden ist, um ein zweites Integralsignal zu erzeugen, das dem Zeitintegral des zweiten Schalterausgangssignals entspricht, und wobei der dritte algebraische Summierer außerdem mit dem zweiten Integrator verbunden ist, um das dritte algebraische Summensignal zu erzeugen, das der Summe aus (1) dem Referenzdrehzahlsignal, (2) dem ersten Integralsignal und (3) dem

zweiten Integralsignal entspricht.

13. Automatische Kupplungssteuerung nach Anspruch 1, bei der:
der Referenzdrehzahlgenerator mit dem Motordrehzahlsensor verbunden ist und das Referenzdrehzahlsignal erzeugt, das dem Motordrehzahlsignal entspricht, und bei dem der Kompensator das Kupplungseinrücksignal für das vollständige Einrücken der Reibkupplung innerhalb eines festgelegten Zeitintervalls nach beginnendem teilweisem Einrücken erzeugt.

14. Automatische Kupplungssteuerung nach Anspruch 1, bei der:
der Kompensator ein Kupplungseinrücksignal erzeugt, das die gewünschte Kupplungsposition kennzeichnet und bei der der Kupplungsaktuator die Position der Reibkupplung entsprechend der gewünschten Kupplungsposition steuert, die durch das Kupplungseinrücksignal gekennzeichnet ist.

15. Automatische Kupplungssteuerung nach Anspruch 1, bei der:
der Kompensator das Kupplungseinrücksignal erzeugt, das durch den gewünschten Kupplungsdruck angezeigt wird, und bei der der Kupplungsaktuator den Druck der Reibkupplung entsprechend dem gewünschten Kupplungsdruck steuert, der durch das Kupplungseinrücksignal charakterisiert ist.

16. Automatische Kupplungssteuerung nach Anspruch 1, wobei die automatische Kupplungssteuerung außerdem eine Drossel aufweist, um das von dem Motor erzeugte Drehmoment zu kontrollieren, sowie einen Drosselsensor, der mit der Drossel verbunden ist, um ein Drosselsignal zu erzeugen, das die Drosselposition kennzeichnet, wobei die Steuereinheit außerdem aufweist:
den Referenzdrehzahlgenerator, der mit dem Motordrehzahlsensor und dem Drosselsensor verbunden ist, um das Referenzdrehzahlsignal entsprechend dem Motordrehzahlsignal und dem Drosselsignal zu erzeugen.

17. Automatische Kupplungssteuerung nach Anspruch 16, bei der:
der Referenzdrehzahlgenerator das Referenzdrehzahlsignal wie folgt erzeugt

$$S_{ref} = E_{sp} \frac{T}{T_{ref}},$$

wobei: S_{ref} das Referenzdrehzahlsignal ; E_{sp} das Motordrehzahlsignal; T das Drosselsignal; und T_{ref} eine Drosselreferenzkonstante ist, die bei einer festgelegten Drosselposition gleich dem Drosselsignal ist.

18. Automatische Kupplungssteuerung nach Anspruch 1, wobei die automatische Kupplungssteuerung außerdem eine Drossel zur Steuerung des von dem Motor erzeugten Drehmoments, einen mit der Drossel verbundenen Drosselsensor zur Erzeugung eines Drosselsignals, das die Drosselposition kennzeichnet, aufweist, wobei die Steuereinheit aufweist:
den Referenzdrehzahlgenerator, der außerdem mit der Drossel verbunden ist und einen Wähler (61) zum Anfahren/ Kriechen aufweist, der mit dem Drosselsensor verbunden ist, um anhand der Größe des Drosselsignals entweder eine Anfahrbetriebsart oder eine Kriechbetriebsart zu wählen, mit einem Kriechdrehzahlreferenzgenerator (62), der mit dem Motordrehzahlsensor und dem Drosselsensor verbunden ist, um ein Kriechdrehzahlreferenzsignal entsprechend dem Motordrehzahlsignal und dem Drosselsignal zu erzeugen, und mit einem Betriebsartenwahlschalter (63), der mit dem Motordrehzahlsensor, dem Wähler für das Anfahren/Kriechbetrieb und dem Kriechdrehzahlreferenzgenerator verbunden ist, um selektiv ein Referenzdrehzahlsignal zu erzeugen, das (1) dem Motordrehzahlsignal entspricht, wenn die Anfahrbetriebsart ausgewählt ist, und das (2) dem Kriechdrehzahlreferenzsignal entspricht, wenn die Kriechbetriebsart ausgewählt ist.

19. Automatische Kupplungssteuerung nach Anspruch 18, bei der:
die Anfahr/Kriechauswahl die Anfahrbetriebsart auswählt, wenn das Drosselsignal eine Drosselposition anzeigt, die größer ist als eine vorbestimmte Drosselposition und ansonsten die Kriechbetriebsart auswählt.

20. Automatische Kupplungssteuerung nach Anspruch 19, bei der:
die festgelegte Drosselposition des Anfahr/Kriechbetriebsartenwählers 25% von Vollgas beträgt.

21. Automatische Kupplungssteuerung nach Anspruch 18, bei der:
der Kriechdrehzahlreferenzgenerator das Kriechdrehzahlreferenzsignal wie folgt bestimmt

$$S_{ref} = E_{sp} \frac{T}{T_{ref}},$$

wobei: S_{ref} das Kriechdrehzahlreferenzsignal; E_{sp} das Motordrehzahlsignal; T das Drosselsignal; und T_{ref} eine Drosselreferenzkonstante ist, die für die festgelegte Drosselposition gleich dem Drosselsignal ist.

22. Automatische Kupplungssteuerung nach einem der Ansprüche 1 bis 21, bei der die Steuereinheit durch diskrete Differenzengleichungen implementiert ist, die von einem Mikrocontroller ausgeführt werden, und:

das Vorfilter und den Kompensator beinhaltet, die als diskrete Differenzengleichungen implementiert sind.

23. Automatische Kupplungssteuerung nach einem der Ansprüche 1 bis 21, bei der ein Getriebe mit einer Eingangswelle, die mit der Ausgangswelle der Reibkupplung verbunden ist, zu der Ausgangswelle hin eine auswählbare Untersetzung liefert, und bei dem eine Getriebeschaltsteuerung mit dem Getriebe verbunden ist, um die von dem Getriebe festgelegte Untersetzung zu steuern, wobei:

die Steuereinheit durch Gleichungen mit endlichen Differenzen implementiert ist, die durch einen Mikrocontroller ausgeführt werden, und einen Koeffizientenspeicher zur Speicherung mehrerer Sets von Koeffizienten aufweist, wobei ein Satz von Koeffizienten jeweils einer auswählbaren Untersetzung des Getriebes entspricht, wobei das Vorfilter in diskreten Differenzgleichungen implementiert ist, die einen Koeffizientensatz nutzen, der aus dem Koeffizientenspeicher aufgerufen worden ist und der Untersetzung des Getriebes entspricht, und wobei der Kompensator in diskreten Differenzengleichungen implementiert ist, die einen Koeffizientensatz nutzen, der aus dem Koeffizientenspeicher aufgerufen worden ist und der Untersetzung des Getriebes entspricht.

24. Verfahren zum automatischen Steuern einer Reibkupplung, wobei die Reibkupplung von einem Controller gesteuert ist und wobei die Reibkuplung (20) eine Eingangswelle (15), die mit einem Motor (10) verbunden ist, und einen mit einer Eingangswelle (25) verbundenen Ausgang aufweist, wobei wenigstens ein schwungbelastetes Antriebsrad (51) mit der Eingangswelle verbunden ist, die ein Drehverhalten aufweist, das auf Drehmomenteingangssignale mit einer Schwingung reagiert, wobei der Controller einen Motordrehzahlsensor (13), der mit dem Motor verbunden ist, enthält, um ein der Drehzahl des Motors entsprechendes Motordrehzahlsignal zu erzeugen, mit einem Getriebeeingangswellendrehzahlsensor (31), der mit der Eingangswelle verbunden ist, um ein der Drehzahl der Eingangswelle entsprechende Getriebeeingangswellendrehzahlsignal zu erzeugen, mit einem Kupplungsaktuator (27), der mit der Reibkupplung verbunden ist, um das Einrücken der Reibkupplung aus dem getrennten bis zu dem vollständig eingekuppelten Zustand gemäß einem Kupplungseinrücksignal zu steuern, und mit einer Steuereinheit (64-78) des Kupplungsaktuators, wobei das Verfahren dadurch gekennzeichnet ist, dass mittels eines Referenzdrehzahlgenerators (61-63) ein Referenzdrehzahlsignal erzeugt wird, das mit dem Motordrehzahlsignal in Verbindung steht, und wobei die Steuereinheit mit dem Referenzdrehzahlgenerator, dem Getriebeeingangswellendrehzahlsensor und dem Kupplungsaktuator verbunden ist, und dass mittels eines Vorfilters (68), das mit dem Referenzdrehzahlgenerator verbunden ist, ein gefiltertes Referenzdrehzahlsignal erzeugt wird, das mittels des ersten algebraischen Summierers (69) das vorgefilterte Referenzdrehzahlsignal und das Getriebeeingangswellendrehzahlsignal algebraisch summiert werden, dass das Ergebnis an einen Kompensator (70) ausgegegen wird, wobei der Kompensator ein kompensiertes Signal abgibt, dass das kompensierte Signal in einen zweiten algebraischen Summierer (71) eingegeben wird, wobei der zweite algebraische Summierer (71) die Eingangssignale des Motordrehzahldifferenzialsignals des Differentiators (73) und die integrierten Motordrehzahlsignale von dem Integrator (74) addiert, um ein Einkuppelsignal auszugeben, um das Einrücken der Reibkupplung in einer Weise zu kontrollieren, die das Getriebeeingangswellendrehzahlsignal veranlasst, sich asymptotisch an das Referenzdrehzahlsignal anzunähern.

25. Verfahren nach Anspruch 24, bei dem:

die Schwingantwort auf Drehmomenteingangssignale des wenigstens einen schwungbelasteten Antriebsrads mittels eines Kompensators vermindert wird, wobei der Kompensator eine Übertragungsfunktion mit einer Bandsperre mit einem Frequenzband in dem Bereich der erwarteten Schwingreaktion auf Drehmomenteingangssignale des wenigstens einen schwungbelasteten Antriebsrads aufweist.

26. Verfahren nach Anspruch 24, bei dem:

der Kompensator den Schleifengewinn erhöht und eine verminderte Empfindlichkeit des Controllers auf auf Veränderungen in der Reaktion auf Drehmomenteingangssignale des wenigstens einen schwungbelasteten Antriebsrads durch eine Übertragungsfunktion beibehält, die einen Bereich erhöhter Verstärkung in dem Frequenzbereich aufweist, in dem die erwartete Reaktion auf Drehmomenteingangssignale des wenigstens einen schwung-

belasteten Antriebsrads ein Minimum ist.

Revendications

1. Dispositif de commande d'un embrayage automatique pour un embrayage à friction (20) possédant un arbre primaire (15) relié à un moteur (10) et une sortie reliée à l'arbre primaire (25), et au moins une roue motrice (51) chargée par inertie reliée à l'arbre primaire avec un comportement en torsion présentant une réponse oscillatoire à des entrées de couple, le dispositif de commande comprenant un capteur de vitesse (13) du moteur relié au moteur, destiné à générer un signal de vitesse du moteur correspondant à la vitesse de rotation du moteur, un capteur de vitesse (31) d'entrée de la transmission relié à l'arbre primaire, destiné à générer un signal de vitesse d'entrée de la transmission correspondant à la vitesse de rotation de l'arbre primaire, un dispositif d'actionnement (27) de l'embrayage relié à l'embrayage à friction, pour commander la mise en prise de l'embrayage à friction de la position libérée à la position complètement en prise selon un signal de mise en prise de l'embrayage, et une unité de commande (64-78) du dispositif d'actionnement de l'embrayage, lequel dispositif de commande est caractérisé par un générateur (61-63) de vitesse de référence couplé au signal de vitesse du moteur pour générer un signal de vitesse de référence, et l'unité de commande est reliée au générateur de vitesse de référence, au capteur de vitesse d'entrée de la transmission et au dispositif d'actionnement de l'embrayage, et comprend un préfiltre (68) relié au générateur de vitesse de référence afin de générer un signal de vitesse de référence filtré, un premier additionneur algébrique (69) relié au capteur de vitesse d'entrée de la transmission et le préfiltre générant un premier signal de somme algébrique correspondant à la différence entre (1) le signal de vitesse de référence filtré et (2) le signal de vitesse d'entrée de la transmission, et un compensateur (70) relié au premier additionneur algébrique destiné à générer le signal de mise en prise de l'embrayage devant être envoyé au dispositif d'actionnement de l'embrayage pour mettre en prise l'embrayage à friction de telle manière que le signal de vitesse d'entrée de la transmission approche le signal de vitesse de référence sous forme d'asymptote.

2. Dispositif de commande d'embrayage automatique selon la revendication 1, dans lequel : le compensateur assure une fonction de transfert et possède un filtre coupe-bande avec une bande de fréquence située dans la plage de la fréquence attendue de la réponse oscillatoire des entrées de couple d'au moins une des roues motrices chargées par inertie, le compensateur réduisant ainsi la réponse oscillatoire aux entrées de couple d'au moins une des roues motrices chargées par inertie.

3. Dispositif de commande d'embrayage automatique selon la revendication 1, dans lequel : le compensateur assure une fonction de transfert et possède une zone de gain accru dans la plage de fréquence où la réponse attendue aux entrées de couple d'au moins une des roues motrices chargées par inertie est une valeur minimum, le compensateur augmentant ainsi le gain de réinjection et maintenant une sensibilité réduite du dispositif de commande aux variations en réponse aux entrées de couple d'au moins une des roues motrices chargées par inertie.

4. Dispositif de commande d'embrayage automatique selon la revendication 1, dans lequel : le préfiltre est un filtre passe-bas possédant une fréquence de coupure sélectionnée pour apporter une réponse transitoire désirée du signal de vitesse d'entrée de la transmission à une fonction échelonnée dans le signal de vitesse de référence.

5. Dispositif de commande d'embrayage automatique selon la revendication 1, dans lequel l'unité de commande comprend en outre :

   un deuxième additionneur algébrique 64 relié au générateur de vitesse de référence et au capteur de vitesse d'entrée de la transmission pour former un deuxième signal de somme algébrique correspondant à la différence entre (1) le signal de vitesse de référence et (2) le signal de vitesse d'entrée de la transmission, un premier intégrateur 65 relié au deuxième additionneur algébrique pour former un premier signal intégral correspondant à l'intégrale de temps du deuxième signal de somme algébrique, et un troisième additionneur algébrique 67 relié au générateur de vitesse de référence et au premier intégrateur pour former un troisième signal de somme algébrique correspondant à la somme du (1) signal de vitesse de référence et (2) du premier signal intégral, le troisième signal de somme algébrique étant envoyé au préfiltre de sorte que le préfiltre est relié au générateur de vitesse de référence via le troisième additionneur algébrique.

6. Dispositif de commande d'embrayage automatique selon la revendication 5, dans lequel l'unité de commande comprend en outre :

   un deuxième intégrateur 66 relié au premier intégrateur pour former un deuxième signal intégral correspondant à l'intégrale de temps du premier signal intégral, et le troisième additionneur algébrique est en outre relié au

deuxième intégrateur pour former le troisième signal de somme algébrique correspondant à la somme (1) du signal de vitesse de référence, (2) du premier signal intégral et (3) du deuxième signal intégral.

7. Dispositif de commande d'embrayage automatique selon la revendication 1, dans lequel l'unité de commande comprend en outre :
un circuit différenciateur 73 relié au capteur de vitesse du moteur destiné à générer un signal différentiel correspondant à la vitesse de changement du signal de vitesse du moteur, et un quatrième additionneur algébrique 71 relié au compensateur et au circuit différenciateur afin de générer le signal d'actionnement de l'embrayage correspondant à la somme (1) de la sortie du compensateur et (2) du signal différentiel.

8. Dispositif de commande d'embrayage automatique selon la revendication 7, dans lequel l'unité de commande comprend en outre un filtre passe-bas 72 placé entre le capteur de vitesse du moteur et le circuit différenciateur.

9. Dispositif de commande d'embrayage automatique selon la revendication 7, dans lequel l'unité de commande comprend en outre un troisième intégrateur 74 relié au circuit différenciateur pour former un troisième signal intégral correspondant à l'intégrale de temps du signal différentiel, et le quatrième additionneur algébrique est en outre relié au troisième intégrateur et génère le signal d'actionnement de l'embrayage correspondant à la somme (1) de la sortie du compensateur, (2) du signal différentiel et (3) du troisième signal intégral.

10. Dispositif de commande d'embrayage automatique selon la revendication 9, dans lequel l'unité de commande comprend en outre :
un deuxième additionneur algébrique 64 relié au générateur de vitesse de référence et au capteur de vitesse d'entrée de la transmission pour former un deuxième signal de somme algébrique correspondant à la différence entre (1) le signal de vitesse de référence et (2) le signal de vitesse d'entrée de la transmission, un détecteur de seuil 75 relié au circuit différenciateur pour déterminer si le signal différentiel est inférieur à un seuil prédéterminé, un premier commutateur 76 relié au deuxième additionneur algébrique et au détecteur de seuil pour générer une sortie du premier commutateur (1) égale à zéro lorsque le détecteur de seuil détermine que le signal différentiel est inférieur au seuil prédéterminé, et (2) dans le cas contraire correspondant au deuxième signal de somme algébrique, un premier intégrateur relié au premier commutateur pour former un premier signal intégral correspondant à l'intégrale de temps de la sortie du premier commutateur, et un troisième additionneur algébrique relié au générateur de vitesse de référence et au premier intégrateur pour former un troisième signal de somme algébrique correspondant à la somme (1) du signal de vitesse de référence et (2) au premier signal intégral, le troisième signal de somme algébrique étant envoyé au préfiltre de sorte que le préfiltre est relié au générateur de vitesse de référence via le troisième additionneur algébrique.

11. Dispositif de commande d'embrayage automatique selon la revendication 10, dans lequel le seuil prédéterminé du détecteur de seuil est égal à zéro.

12. Dispositif de commande d'embrayage automatique selon la revendication 10, dans lequel l'unité de commande comprend en outre :
un deuxième commutateur 77 relié au détecteur de seuil et au premier intégrateur pour générer une sortie du deuxième commutateur (1) égale à zéro lorsque le détecteur de seuil détermine que le signal différentiel est inférieur au seuil prédéterminé, et (2) dans le cas contraire, correspondant au premier signal intégral, un deuxième intégrateur relié au deuxième commutateur pour former un deuxième signal intégral correspondant à l'intégrale de temps de la sortie du deuxième commutateur, et le troisième additionneur algébrique est en outre relié au deuxième intégrateur pour former le troisième signal de somme algébrique correspondant à la somme (1) du signal de vitesse de référence, (2) du premier signal intégral et (3) du deuxième signal intégral.

13. Dispositif de commande d'embrayage automatique selon la revendication 1, dans lequel : le générateur de vitesse de référence est relié au capteur de vitesse du moteur et génère le signal de vitesse de référence correspondant au signal de vitesse du moteur ; et le compensateur génère le signal de mise en prise de l'embrayage pour mettre complètement en prise l'embrayage à friction dans un intervalle prédéterminé de temps après une mise en prise initiale partielle.

14. Dispositif de commande d'embrayage automatique selon la revendication 1, dans lequel : le générateur génère le signal de mise en prise de l'embrayage indiquant la position désirée d'embrayage ; et le dispositif d'actionnement de l'embrayage commande la position de l'embrayage à friction correspondant à la position désirée d'embrayage indiquée par le signal de mise en prise de l'embrayage.

15. Dispositif de commande d'embrayage automatique selon la revendication 1, dans lequel : le compensateur génère le signal de mise en prise de l'embrayage indiquant la pression désirée d'embrayage ; et le dispositif d'actionnement de l'embrayage commande la pression de l'embrayage à friction correspondant à la pression désirée d'embrayage indiquée par le signal de mise en prise de l'embrayage.

16. Dispositif de commande d'embrayage automatique selon la revendication 1, le dispositif de commande d'embrayage automatique comprenant en outre un papillon des gaz pour commander un couple généré par le moteur et un capteur de papillon des gaz relié au papillon des gaz pour générer un signal de papillon des gaz indiquant la position du papillon des gaz, l'unité de commande comprenant en outre :

le générateur de vitesse de référence relié au capteur de vitesse du moteur et au capteur du papillon des gaz afin de générer le signal de vitesse de référence correspondant au signal de vitesse du moteur et au signal du papillon des gaz.

17. Dispositif de commande d'embrayage automatique selon la revendication 16, dans lequel :

le générateur de vitesse de référence génère le signal de vitesse de référence suivant :

$$S_{ref} = E_{sp} \frac{T}{T_{ref}}$$

avec: S_{ref} le signal de vitesse de référence ; E_{sp} le signal de vitesse du moteur ; T le signal du papillon des gaz ; et T_{ref} une constante de référence du papillon des gaz égale au signal du papillon des gaz pour une position prédéterminée du papillon des gaz.

18. Dispositif de commande d'embrayage automatique selon la revendication 1, le dispositif de commande d'embrayage automatique comprenant en outre un papillon des gaz pour commander le couple généré par le moteur et un capteur de papillon des gaz relié au papillon des gaz pour générer un signal de papillon des gaz indiquant la position du papillon des gaz, l'unité de commande comprenant :

le générateur de vitesse de référence, lequel est en outre relié au papillon des gaz et comprend un sélecteur 61 de lancement/vitesse lente relié au capteur de papillon des gaz pour sélectionner soit un mode de lancement soit un mode de vitesse lente en fonction de l'importance du signal du papillon des gaz, un générateur de référence 62 de vitesse lente relié au capteur de vitesse du moteur et au capteur du papillon des gaz destiné à générer un signal de référence de vitesse lente correspondant au signal de vitesse du moteur et au signal du papillon des gaz, et un commutateur 63 de sélection de mode relié au capteur de vitesse du moteur, au sélecteur de lancement/vitesse lente et au générateur de référence de vitesse lente pour générer de manière sélective un signal de vitesse de référence correspondant (1) au signal de vitesse du moteur si le mode de lancement est sélectionné et (2) au signal de référence de vitesse lente si le mode vitesse lente est sélectionné.

19. Dispositif de commande d'embrayage automatique selon la revendication 18, dans lequel :

la sélection lancement/vitesse lente sélectionne le mode de lancement si le signal du papillon des gaz indique une position du papillon des gaz supérieure à une position prédéterminée du papillon des gaz ; dans le cas contraire, il sélectionne le mode de vitesse lente.

20. Dispositif de commande d'embrayage automatique selon la revendication 19, dans lequel :

la position prédéterminée du papillon des gaz du sélecteur de lancement/vitesse lente correspond à 25 % de l'ouverture complète du papillon.

21. Dispositif de commande d'embrayage automatique selon la revendication 18, dans lequel :

le générateur de référence de vitesse lente génère le signal de référence de vitesse lente suivant :

$$S_{ref} = E_{sp} \frac{T}{T_{ref}}$$

avec : S_{ref} le signal de référence de vitesse lente ; E_{sp} le signal de vitesse du moteur ; T le signal de papillon des gaz ; et T_{ref} une constante de référence du papillon des gaz égale au signal du papillon des gaz pour une position prédéterminée du papillon des gaz.

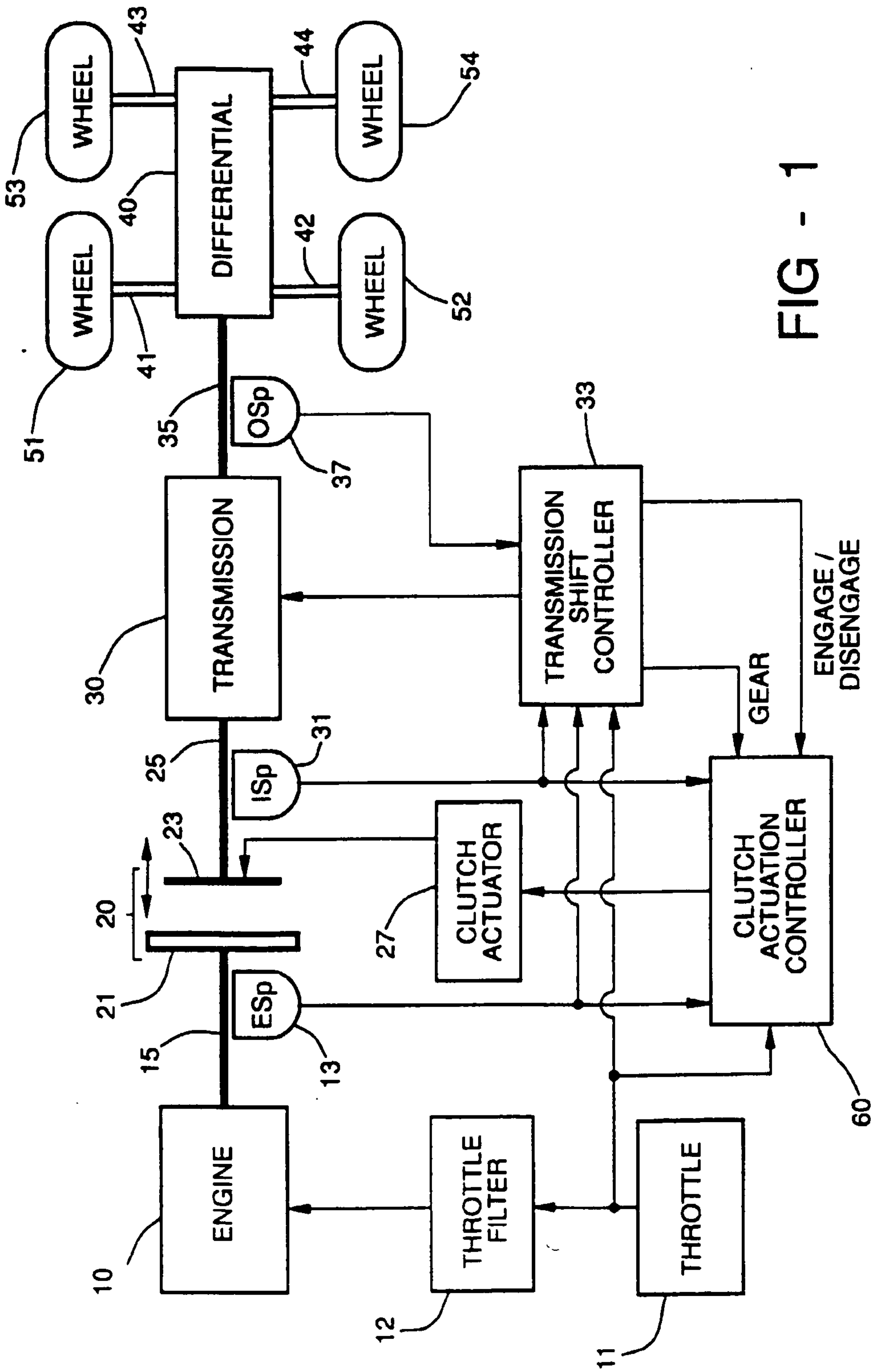
22. Dispositif de commande d'embrayage automatique selon l'une quelconque des revendications 1 à 21, dans lequel l'unité de commande est mise en oeuvre par des équations différentielles discrètes exécutées par un microcontrôleur et comprend :
le préfiltre et le compensateur mis en oeuvre dans des équations différentielles discrètes.

23. Dispositif de commande d'embrayage automatique selon l'une quelconque des revendications 1 à 21, dans lequel une transmission possédant un arbre primaire relié à un arbre principal de l'embrayage à friction fournit un rapport de vitesse susceptible d'être sélectionné sur l'arbre principal, et un dispositif de commande du changement de vitesse relié à la transmission pour commander le rapport de vitesse sélectionné par la transmission, dans lequel :
l'unité de commande est mise en oeuvre via des équations différentielles discrètes exécutées par un microcontrôleur et comprenant une mémoire des coefficients pour stocker une pluralité d'ensembles de coefficients, un ensemble de coefficients correspondant à chaque rapport de boîte susceptible d'être sélectionné de la transmission, le préfiltre est mis en oeuvre dans des équations différentielles discrètes faisant appel à un ensemble de coefficients retirés de ladite mémoire des coefficients correspondant au rapport de vitesse de la transmission, et le compensateur est mis en oeuvre dans des équations différentielles discrètes faisant appel à un ensemble de coefficients retirés de ladite mémoire des coefficients correspondant au rapport de vitesse de la transmission.

24. Procédé pour commander automatiquement un embrayage à friction, l'embrayage à friction étant'commandé par un dispositif de commande et l'embrayage à friction (20) possédant un arbre primaire (15) relié à un moteur (10) et une sortie reliée à un arbre primaire (25), et au moins une roue motrice (51), chargée par inertie reliée à l'arbre primaire possédant un comportement en torsion présentant une réponse oscillatoire à des entrées de couple, le dispositif de commande comprenant un capteur de vitesse (13) du moteur, relié au moteur, destiné à générer un signal de vitesse du moteur correspondant à la vitesse de rotation du moteur, un capteur de vitesse (31) d'entrée de la transmission relié à l'arbre primaire destiné à générer un signal de vitesse d'entrée de la transmission correspondant à la vitesse de rotation de l'arbre primaire, un dispositif d'actionnement (27) de l'embrayage relié à l'embrayage à friction pour commander la mise en prise de l'embrayage à friction de la position libérée à la position complètement en prise selon un signal de mise en prise de l'embrayage, et une unité de commande (64-78) du dispositif d'actionnement de l'embrayage, lequel procédé est caractérisé par la génération d'un signal de vitesse de référence par l'intermédiaire d'un générateur (61-63) de vitesse de référence couplé au signal de vitesse du moteur, et l'unité de commande étant reliée au générateur de vitesse de référence, au capteur de vitesse d'entrée de la transmission et au dispositif d'actionnement de l'embrayage et générant un signal de vitesse de référence filtré à l'aide d'un préfiltre (68) relié au générateur de vitesse de référence en additionnant de manière algébrique, au moyen d'un premier additionneur algébrique (69), le signal de vitesse de référence préfiltré et le signal de vitesse d'entrée de la transmission, en'envoyant le résultat à un compensateur (70), ledit compensateur émettant un signal compensé, et envoyant ledit signal compensé au deuxième additionneur algébrique (71), le deuxième additionneur algébrique (71) additionnant les signaux d'entrée à partïr des signaux de vitesse différentiels du moteur en provenance du circuit différenciateur (73), et les signaux intégrés de vitesse du moteur en provenance d'un intégrateur (74) pour émettre un signal de mise en prise de l'embrayage pour commander la mise en prise de l'embrayage à friction de telle manière que le signal de vitesse d'entrée de la transmission approche le signal de vitesse de référence sous forme d'asymptote.

25. Procédé selon la revendication 24, dans lequel : la réponse oscillatoire aux entrées de couple d'au moins une des roues chargées par inertie est réduite à l'aide d'un compensateur, le compensateur assure une fonction de transfert et possède un filtre coupe-bande avec une bande de fréquence située dans la plage de la fréquence attendue de la réponse oscillatoire des entrées de couple d'au moins une des rques motrices chargée par inertie.

26. Procédé selon la revendication 24, dans lequel :
le compensateur augmente le gain de réinjection et maintient une sensibilité réduite du dispositif de commande aux variations de réponse aux entrées de couple d'au moins une des roues motrices chargées par inertie par l'intermédiaire d'une fonction de transfert possédant une zone de gain accru dans la plage de fréquence où la réponse attendue aux entrées de couple d'au moins une des roues motrices chargées par inertie est une valeur minimale.


WHEEL
WHEEL
WHEEL
WHEEL
DIFFERENTIAL
TRANSMISSION
OSp
ISp
ESp
TRANSMISSION SHIFT CONTROLLER
ENGAGE / DISENGAGE
GEAR
CLUTCH ACTUATOR
CLUTCH ACTUATION CONTROLLER
ENGINE
THROTTLE FILTER
THROTTLE
10
11
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13
15
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54
60

FIG - 1

FIG - 2

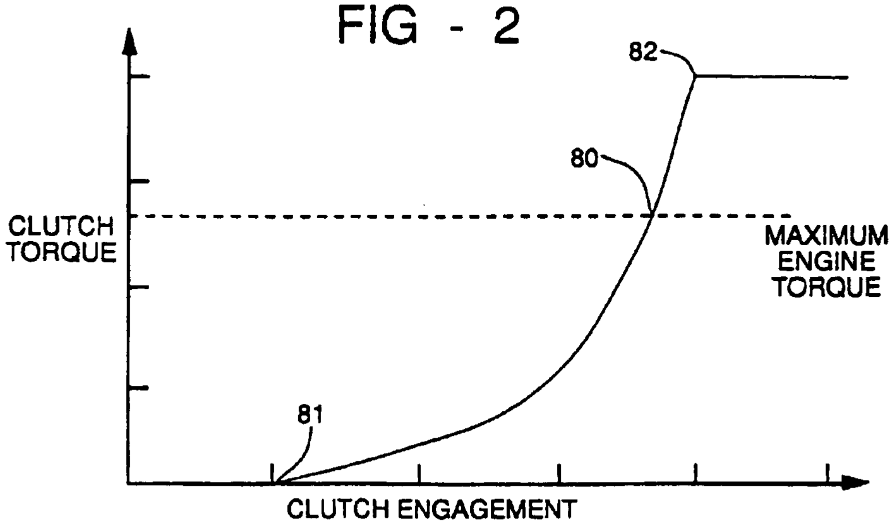


FIG - 3

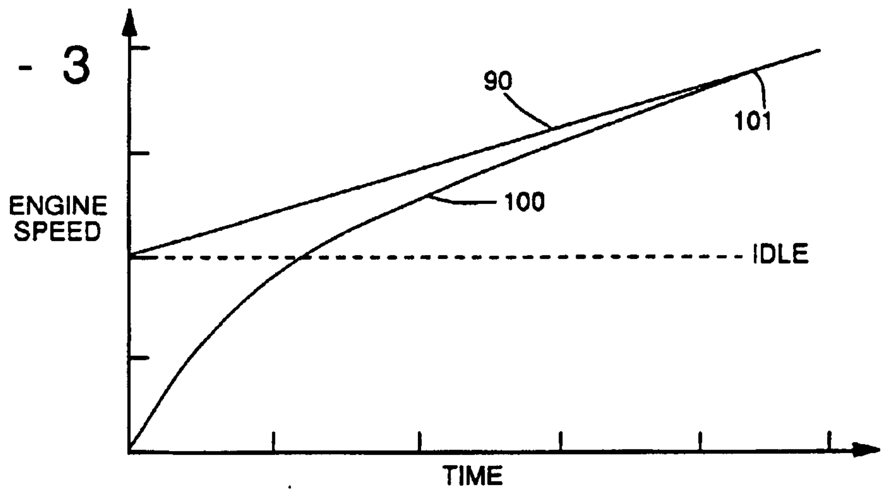


FIG - 4

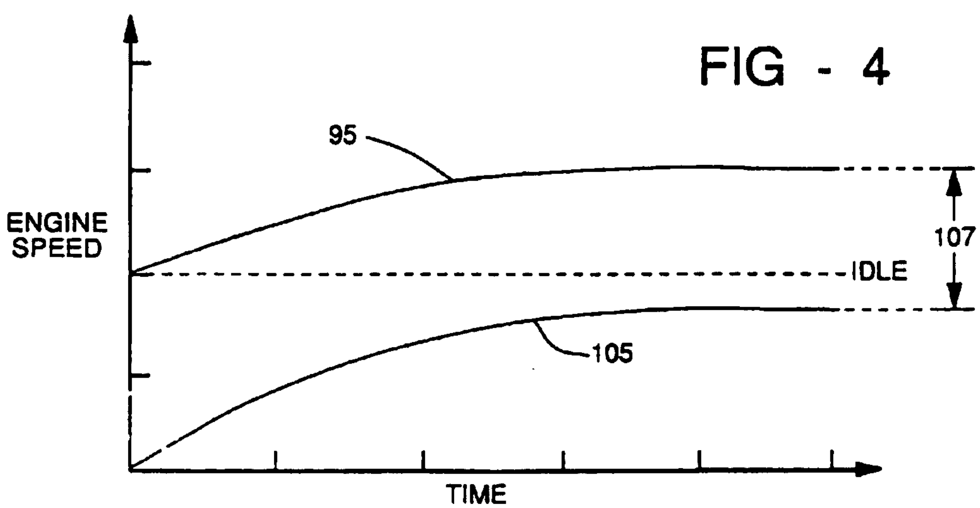
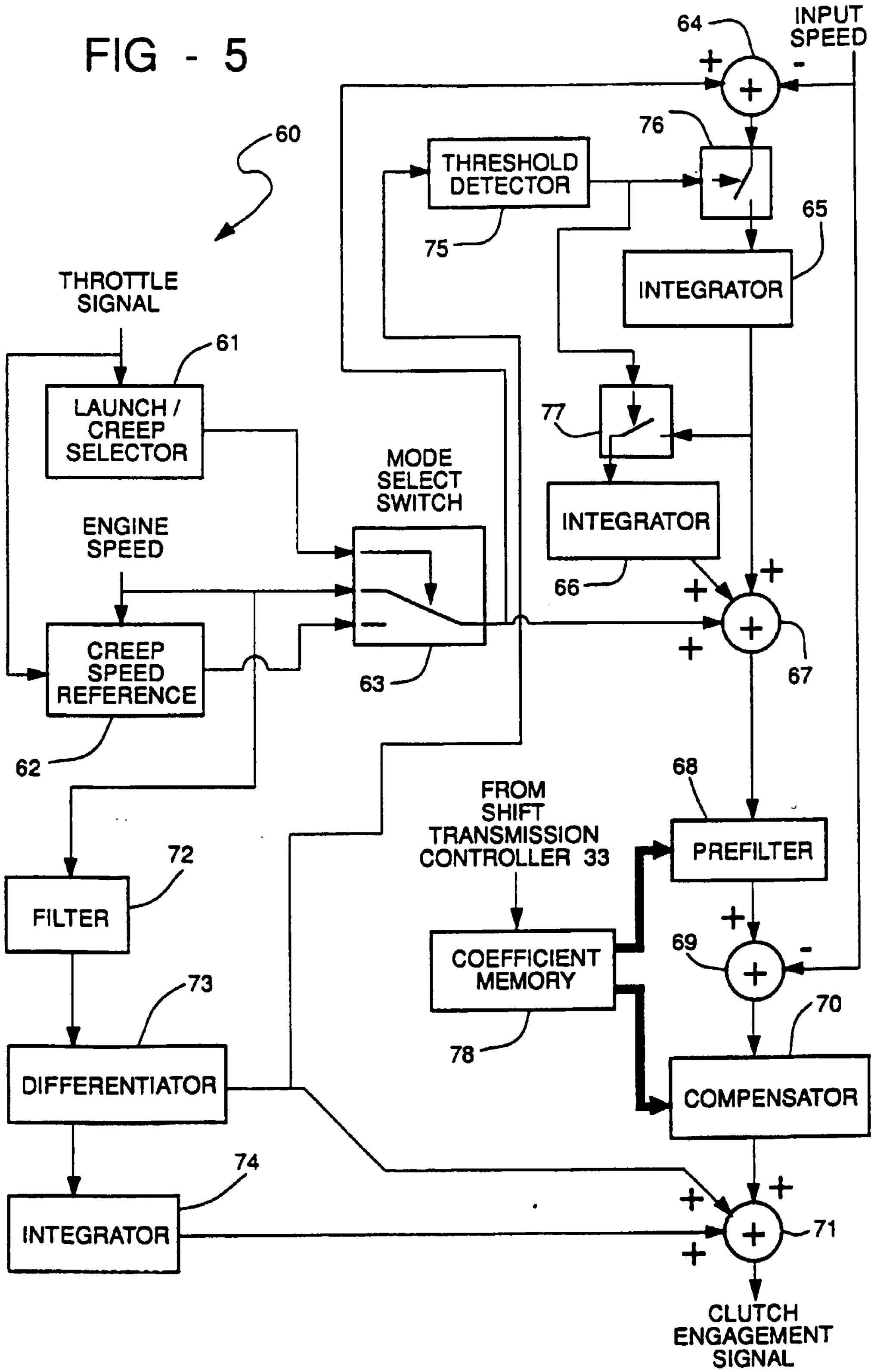


FIG - 5


60
THROTTLE SIGNAL
61
LAUNCH / CREEP SELECTOR
ENGINE SPEED
CREEP SPEED REFERENCE
62
MODE SELECT SWITCH
63
THRESHOLD DETECTOR
75
64
INPUT SPEED
76
65
INTEGRATOR
77
INTEGRATOR
66
67
68
PREFILTER
FROM SHIFT TRANSMISSION CONTROLLER 33
COEFFICIENT MEMORY
78
69
70
COMPENSATOR
FILTER
72
73
DIFFERENTIATOR
74
INTEGRATOR
71
CLUTCH ENGAGEMENT SIGNAL